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DESIGN, ANALYSIS AND OPTIMIZATION OF A MULTI-TUBULAR SPACE FRAME

SURAJ ARU[1], PRAVIN JADHAV[2], VINAY JADHAV[3], AKOOL KUMAR[4] & PRATIM ANGANE[5]

[1]Department of Production Engineering & Industrial Management, College of Engineering, Pune, Maharashtra, India

[2,3,4,5] Department of Mechanical Engineering, College of Engineering, Pune, Maharashtra, India

ABSTRACT

Multi tubular space frames, often referred to as roll-cage acts as a structural embody for various types of automotive vehicles. A space frame serves a dual function of giving structural safety to the vehicle and at the same time incorporates the different subsystems like suspension, steering, transmission on to it. This makes the space frame a very vital and most cautious designed component, especially in case of vehicles like race cars and All-Terrain Vehicles. In order to ensure maximum safety along with due consideration to the weight aspect, the roll-cage of an All-Terrain Vehicle is designed. To fulfil these criteria it is important to consider various parameters involved in the design of a roll-cage, right from the material to be used up to the forces and impacts that it might encounter. Through this study, we aimed to design, analyse and optimize a roll-cage so as to achieve the target of apt strength to weight ratio. With the help of MATLAB, CAD modelling software’s and ANSYS workbench 14.0, the model was designed and optimised to serve the desired purpose. Material was selected after conducting an extensive market survey and on the basis of wetted point method. This sequential approach was adopted for the roll-cage design of BAJA vehicle and proved to be effective.

KEYWORDS: Multi-Tubular Space Frames, Roll Cage, Analysis, FEA, ANSYS 14.0, Optimization

INTRODUCTION

Multi-Tubular Space frame is a structure which consists of tubes of different cross sections that supports various components of a vehicle and at the same time protects the passengers. It is commonly incorporated for All Terrain Vehicles - ATV’s.

The design discussed in the following paper is of a Multi-Tubular Space frame all-terrain vehicle powered by a 300cc engine. Since the vehicle is powered by 300cc engine, weight becomes quite a crucial factor which decides the speed of the vehicle. To have the maximum acceleration, there arises a need to reduce the weight of the car and under these circumstances; optimization for the strength to weight ratio of the car becomes very important. To best optimize this balance, the use of 3D modelling and finite element analysis (FEA) software is extremely useful in addition to conventional analysis. The following paper discusses the techniques to optimize the strength of the chassis. It will cover the design constraints required by SAE, the material selection, initial design, structural analysis and design modifications. It will finally cover the results of the actual real world usage of the frame design. To have a better acceleration, it is always recommended to keep the mass of the vehicle as low as possible. In race car vehicle dynamics, sprung mass plays an important role. So, it becomes the prime task of the engineer to come up with a compromise between strength, reliability and weight which have let them explore the field of optimisation in more detail.

TERMINOLOGY

The design of the space frame/roll-cage mainly revolves around other subsystems, as it has to accommodate them. The roll-cage design is governed by a set of rules mentioned in the rulebook provide by SAE. The rules define the space frame based on geometry as well as material. On the basis of geometry, the members of the space frame are further classified as primary and secondary members. Different members of the space frame are denoted by various annotations which are mentioned below.

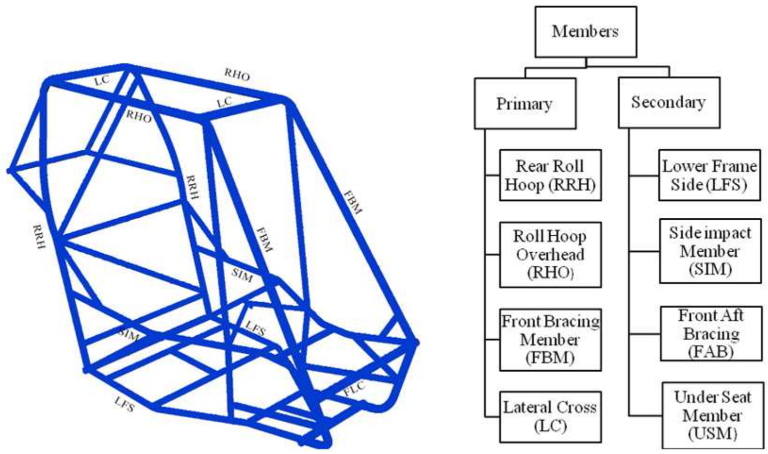

Figure 1: Roll-Cage Terminology

MATERIAL SELECTION PARAMETERS

The overall weight of the vehicle is a crucial factor as the frame is used in a racing vehicle and so must be given due importance. An optimum balance of fulfilling the design requirements and minimizing the weight is crucial for a successful design. A careful analysis for material selection plays an important role in achieving this target.

The most commonly used materials for the roll-cage are AISI 1018 & AISI 4130.The following table compares the three materials

Table 1: 1018 vs. 4130

Material	AISI 1018	AISI 4130
Yield Strength	417 MPa	709 MPa
Ultimate Strength	473 MPa	810 MPa
Bending Strength	402.9Nm	415Nm
Welding Type	MIG Welding	TIG Welding
Availability	Easily available in India	Not available in India
Cost	Less costly	Expensive

From the above table it can be deduced that AISI 4130 has a much better strength to weight ratio. Also, by using AISI 4130, we can ensure a straight weight reduction of 17% per meter tubing length of the space frame without compromising on its strength.

Selection of Cross-Sections

Primary Members

To select the most appropriate section for primary members, an analysis was done for bending strength, bending stiffness and weight per metre of the cross-section. The graphs of Bending Strength and Bending Stiffness versus wall

thickness were plotted in MATLAB. They also include the minimum requirements for bending stiffness, bending strength and minimum wall thickness (1.6 mm) as specified by the rulebook. The cross sections shortlisted were 1 inch, 1.125 inch and 1.25 inch. Through a market survey, it was found that the above mentioned pipe diameters were available in thicknesses available were 1mm, 1.2mm, 1.6mm, 1.8mm and 2.1mm.

Bending Strength is given by = $\frac{(S_y X 1)}{C}$ Bending Stiffness is given by = $E \times 1$

Where, S_y =Yield strength (MPa) Where, E = Modulus of Elasticity

I = Second moment of area I = Second moment of area

C = Distance of extreme most fibre from the neutral axis

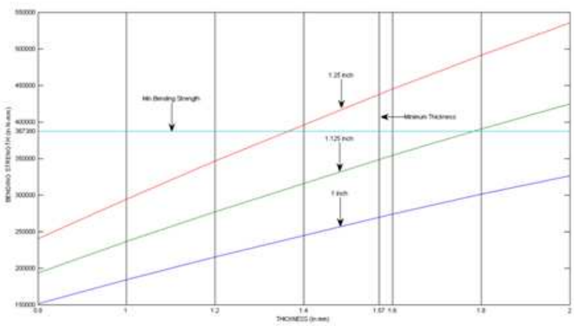

Figure 2: Bending Strength v/s Thickness

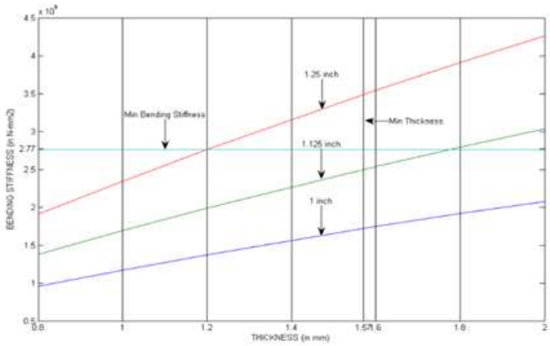

Figure 3: Bending Stiffness v/s Thickness

Table 2: Comparison of Cross-Sections

Cross-Section	Weight Per Metre
1.125inch x 1.8 mm	1.190191 kg
1.25 inch x 1.6 mm	1.191302 kg
1.25 inch x 1.8 mm	1.331324 kg

Referring the graphs, the cross-sections satisfying the minimum requirements for bending strength, bending stiffness are 1.125inch x 1.8 mm, 1.25 inch x 1.6 mm and 1.25 inch x 1.8 mm.

Since the cross section 1.125 inch x 1.8 mm gives minimum weight along with satisfying all the conditions, it was selected for the primary members.

Secondary Members

Secondary members are provided with an aim of providing structural support to the entire frame and primary members in particular. Since these members are specifically structural they are used of smaller diameter and thickness as compared to the primary ones.

CAD MODEL

The entire Space frame was modelled in Pro-Engineer software.

The following are the considerations for the design:

- **Driver Ergonomics:** The emphasis of the design is on driver comfort.

- **Nodal Geometry:** To increase the load transfer path.
- Mounting points for the integration of Suspension, Transmission, Steering and Brakes.

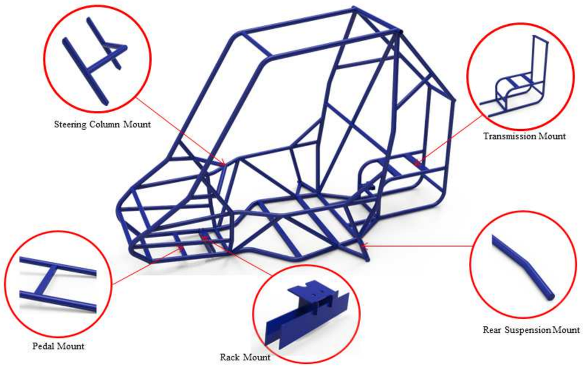

Figure 4: CAD Model of Space Frame under Consideration

FINITE ELEMENT ANALYSIS

The multi tubular space frame of an all-terrain vehicle should be capable of enduring harsh off road environments. Finite element analysis of the roll-cage was done using ANSYS 14.The roll-cage was analysed for various conditions like Front impact, Side impact, and Front roll over, Side roll over, Torsional Stiffness with the main focus on driver's safety. The results were studied and the necessary changes were incorporated in the design wherever necessary.

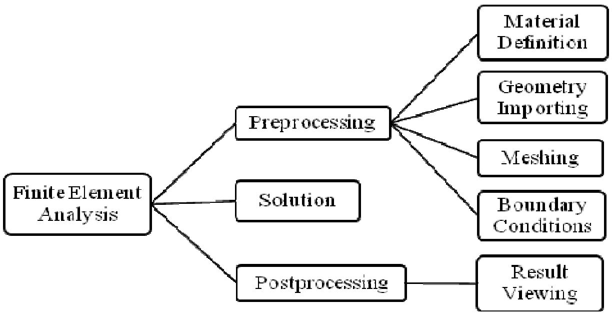

Figure 5

Meshing

The Roll-cage mid surfaces were created in ProE and the .igs file was imported in Ansys Workbench. 2d meshing was carried out since the thickness of the pipe was much less than the diameter of the tube. Shell elements were used for carrying out the 2d meshing of the roll-cage.

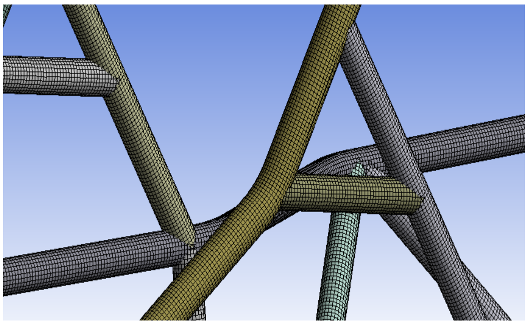

Figure 6: Roll-Cage Meshing

Impact Analysis

The Impact forces were calculated using Newton's second law which states that the net force acting on a body is equal to the product of mass and acceleration of the body.

$$Force = Mass \times Acceleration$$
$$Force = Rate\ of\ change\ in\ momentum$$
$$Impulse = Force \times Time = Change\ in\ momentum = Mass\ \times Change\ in\ velocity$$

- **Front Impact:** In actual conditions, the car is going to hit a tree, another car or a wall. In the first 2 cases, the tree and the other car are deformable bodies. So the time of impact will be greater, around 0.3 seconds, while the wall is considered as non-deformable i.e. a rigid body. Hence the time of impact will be obviously less than that in the above case. It is obvious that the impact force in the case of wall will be more than the first two cases. The vehicle was considered to be moving with a velocity of 45 kmph and time of impact as 0.13 seconds.
- **Side Impact:** Since both bodies involved are deformable, the time of impact is slightly more than that of front impact. In case of side impact, the vehicle was considered to be in a stationary state. Impact was subjected on the side by an identical vehicle at a speed of 45 kmph. Time of impact is taken as 0.3 seconds because both the bodies are deformable.

Front Impact Analysis

The front impact analysis is done for analysing the rigidity of a roll cage as well as the safety of the driver in case of a head on collision of the car.

Table 3

Max Force	28KN (9.8g)
Force Applied on	Front most members
Fixed	Suspension Mounting Points

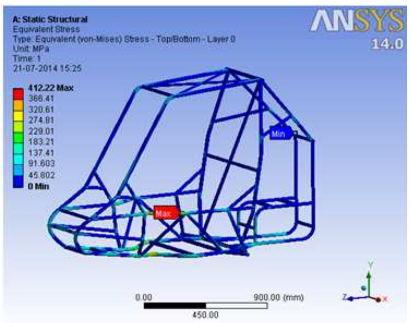

Figure 7: Front Impact Stress Distribution

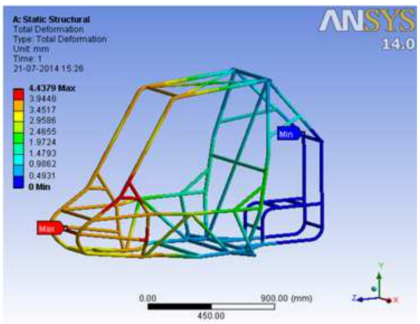

Figure 8: Front Impact Deformation

Table 4

Max Stress	Max Deformation	Factor of Safety
412.22 MPa	4.4379 mm	1.011

Modifications

- Slight changes in the position of SIM to LFS bracing members are done.
- Additional members joining LFS and under seat member.

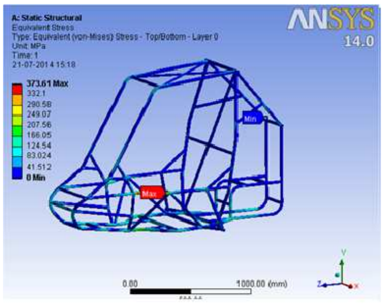

Figure 9: Front Impact Stress Distribution

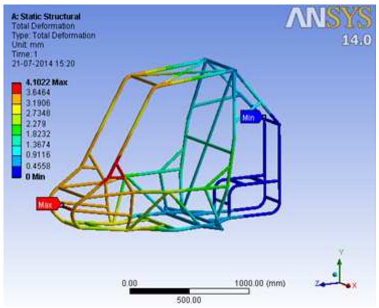

Figure 10: Front Impact Deformation

Table 5

Max Stress	Max Deformation	Factor of Safety
373.61 MPa	4.1022 mm	1.116

Side Impact

Side impact analysis of a vehicle is done to check the strength of the roll-cage in the case of accident involving the vehicle hit by another car from side.

Table 6

Max Force	2.8KN (1.1)g
Force Applied on	Side most members
Fixed	Suspension Mounting Points

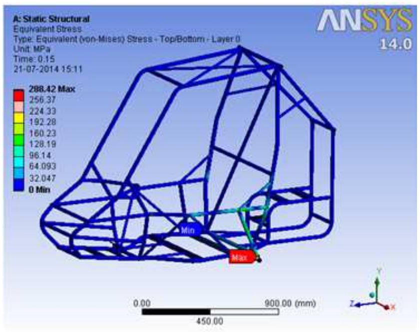

Figure 11: Side Impact Stress Distribution

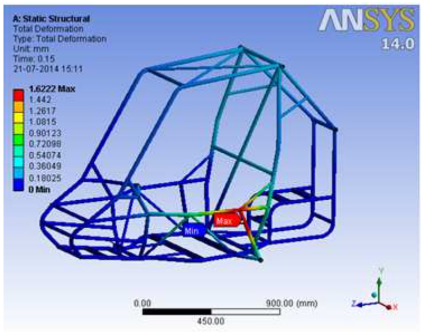

Figure 12: Side Impact Deformation

Table 7

Max Stress	Max Deformation	Factor of Safety
288.42 MPa	1.6222 mm	1.446

ROLL-OVER

Being an off road vehicle the chassis is prone to the situations of roll-over. Two cases of roll over are considered

below i.e. side roll over and front roll over. An off-road vehicle should be checked for the rollover condition by keeping in mind the purpose it is going to serve.

Side Roll Over

This analysis is performed by considering the toppling of the vehicle while sharp cornering. The case might be considered when the vehicle is going around the corner having opposite angle of banking.

Table 8

Max Force	2.8KN (1.1)g
Force Applied on	FBM-RHO Bend

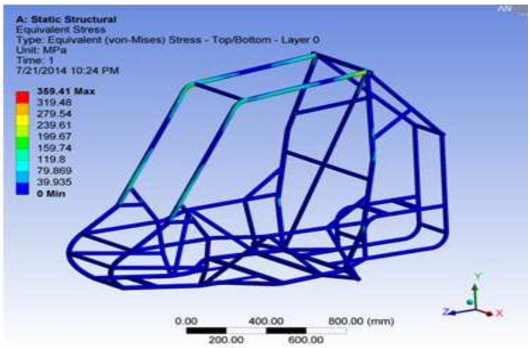

Figure 13: Side Roll over Stress Distribution

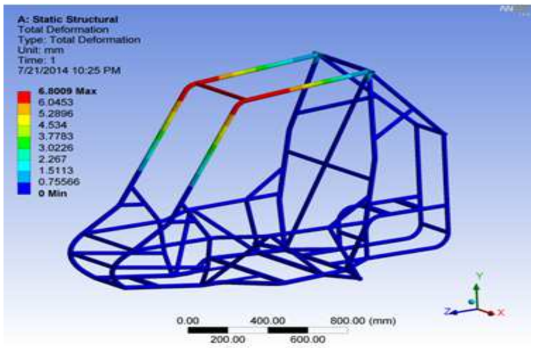

Figure 14: Side Roll Over Deformation

Table 9

Max Stress	Max Deformation	Factor of Safety
359.41 MPa	6.8009 mm	1.16

Modifications

- Cross members joining RHO Members

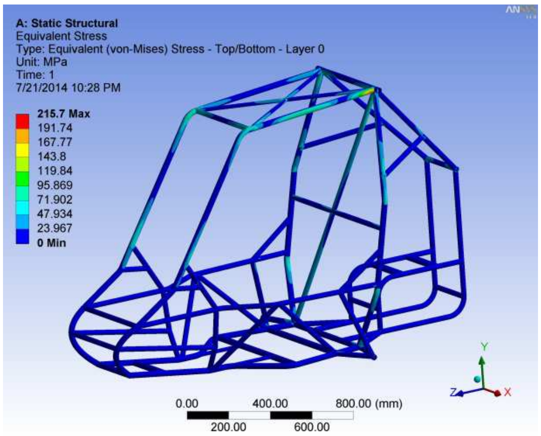

Figure 15: Side Roll Over Stress

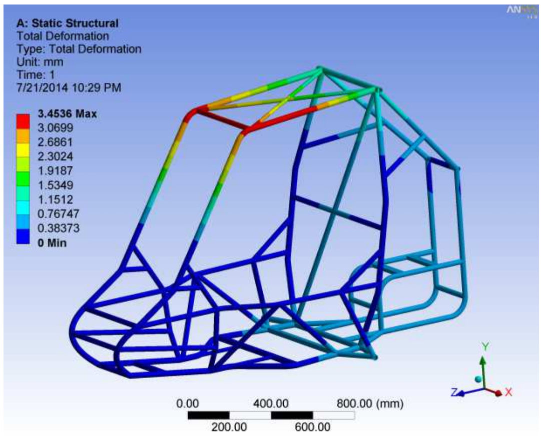

Figure 16: Side Roll Over Deformation

Table 10

Max Stress	Max Deformation	Factor of Safety
215.7 MPa	3.4536 mm	1.933

Front Roll Over

In the case of front roll over, the vehicle is considered as toppling while coming down a hill.

Table 11

Max Force	2.8KN(1.1g)
Force Applied on	At 45° to FBM-RHO Bends
Fixed	Suspension Mounts

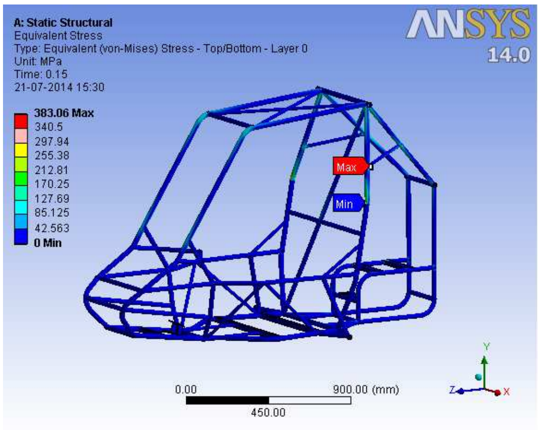

Figure 17: Front Roll Over Stress

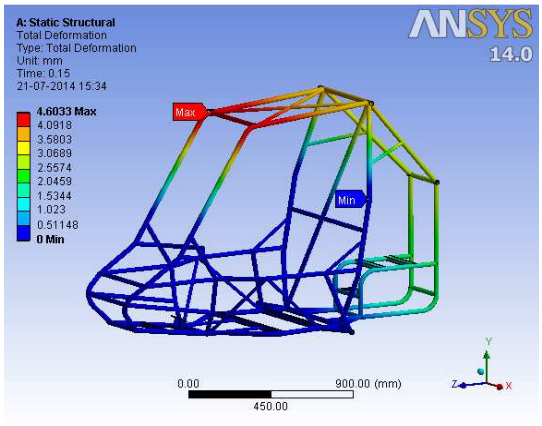

Figure 18: Front Roll Over Deformation

Table 12

Max Stress	Max Deformation	Factor of Safety
383.06 MPa	4.6033 mm	1.088

TORSIONAL STIFFNESS ANALYSIS

The chassis should be stiff enough to sustain dynamic suspension loads. When the vehicle is negotiating the bump there might be a case of alternating bumps on left and right wheels. Considering the left wheel is having the upward travel (jounce) and the right wheel is having the downward travel (rebound) the spring forces will act in the opposite direction composing a couple on front of the vehicle.

This couple tries to produce the torsional stress in the chassis. for the worst case scenario the diagonally opposite wheels are having the opposite wheel travel i.e. front right wheel is having the vertically upward travel and at same time rear left wheel is having the vertically downward travel producing a couple diagonally. This couple is responsible for the torsional stresses in the vehicle.

Table 13

Max Force	Front-1.53KN (3g)	Rear-2.354KN (3g)
Force Applied on	Diagonally Opposite Suspension Mounts	
Fixed	Diagonally Opposite Suspension Mounts	

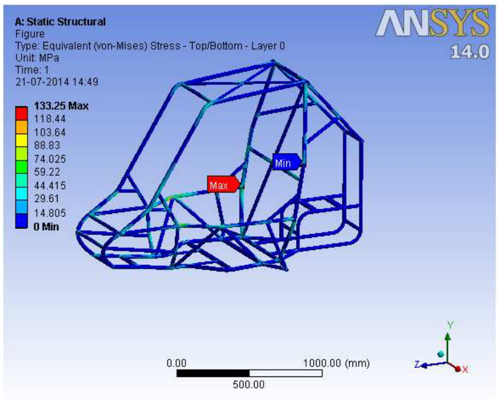

Figure 19: Torsional Stress Distribution

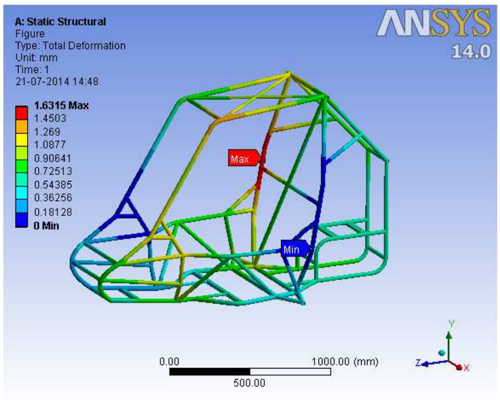

Figure 20: Torsional Deformation

Max Stress	Max Deformation	Factor of Safety
133.25 MPa	1.6315mm	3.129

Torsional Stiffness

The maximum deformation was at the rear suspension mount

F = 2354 N

L = Distance between diagonally opposite suspension mounts=490mm

D = Vertical deformation in suspension mounts

Θ = Angular deformation

Tan(θ)= D/(L/2)

Torsional Stiffness = (F x L) / θ.

D=1.252mm

Torsional Stiffness= 3939.54 Nm/degree

MODAL ANALYSIS

Modal analysis was done to avoid resonance of the roll-cage. Engine is the main source of vibration in the vehicle. Since a 4-stroke single cylinder engine is used, dominant half order excitation frequency of the engine was calculated at idle and maximum rpm.It was concluded from modal analysis that first six modes of vibrating frequency does not lie between working frequency of the engine and hence resonance will not occur. The vibration frequency of the engine ranges from 15Hz to 31.667Hz

Table 14

Mode	Frequency
1	61.37 Hz
2	81.561Hz
3	87.503 Hz
4	108.1 Hz
5	143.5 Hz
6	161.9 Hz

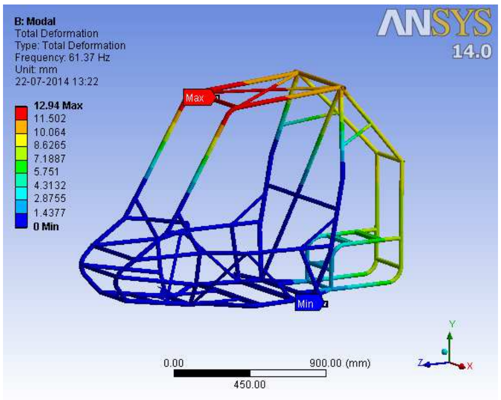

Figure 21: 1st Mode Deformation

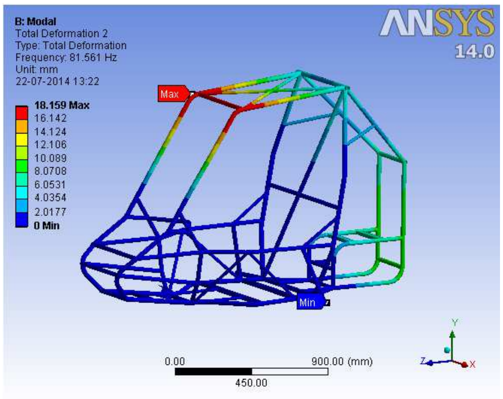

Figure 22: 2nd Mode Deformation

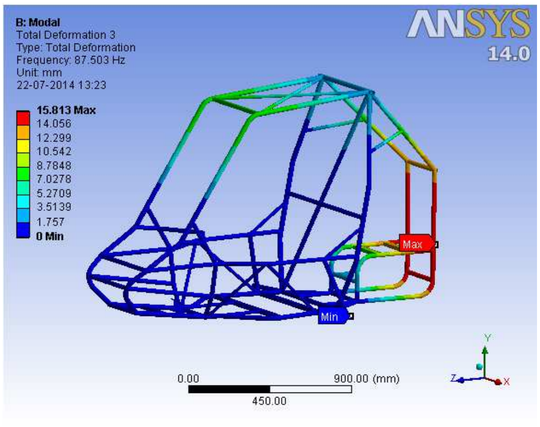

Figure 23: 3rd Mode Deformation

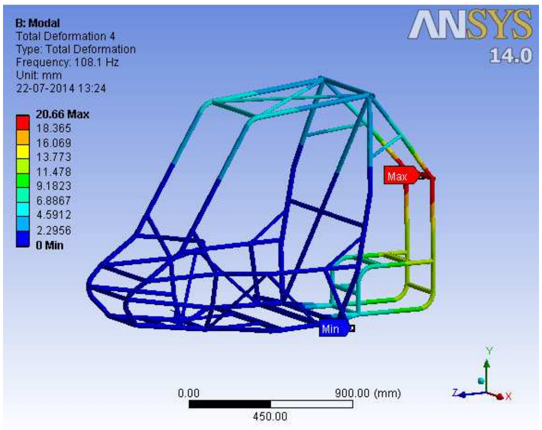

Figure 24: 4th Mode Deformation

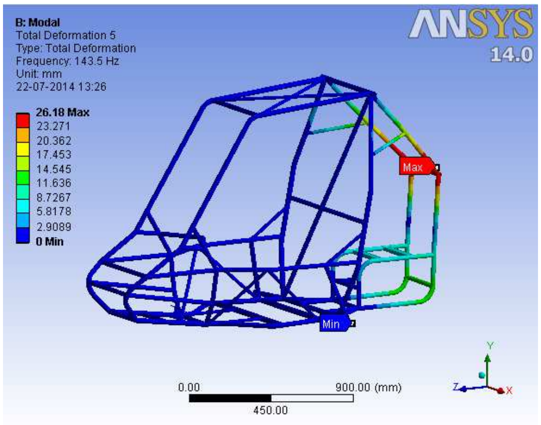

Figure 25: 5th Mode Deformation

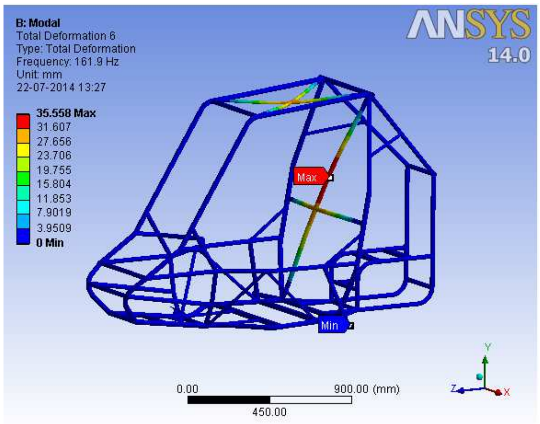

Figure 26: 6th Mode Deformation

CONCLUSIONS ON FEA OF CHASSIS

After performing the Front impact, side impact, and roll over and torsion analyses and making the necessary changes, the following design was finalized.

- The above designed chassis is much stiffer and stronger than the preliminary design. The chassis members were optimized by changing dimensions of the pipes in required positions.

- In the case of front impact and side impact analysis, the deformation of the front most member of the roll cage must be less than 10% of the clearance between driver roll cage members ensuring the safety of the driver. Though the factor of safety in front impact is 1.116 and in side impact is 1.446, deformation is within limit, ensuring that the driver is safe.

- For front roll over, the deformation is important than the maximum stress. The deformation is 4.6033mm and it is safe for the driver.

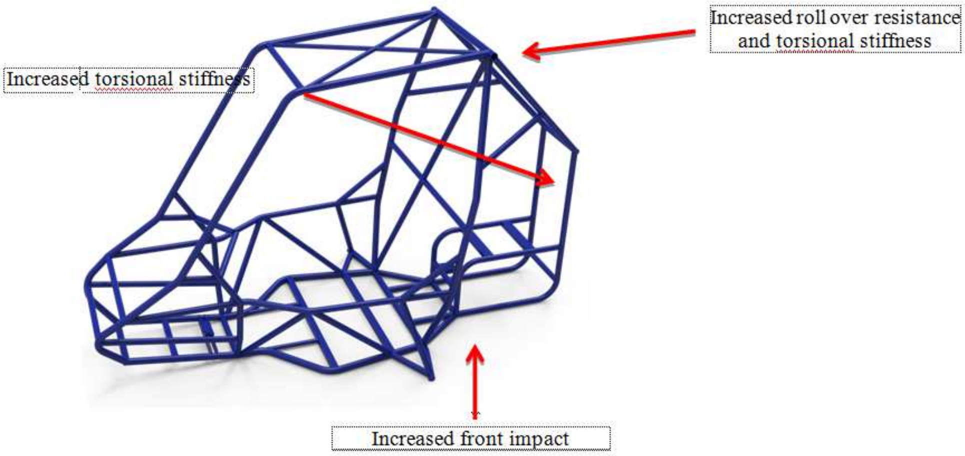

Figure 27: Final Optimised Space Frame

- Usually side roll over analysis is not so significant in case of commercial vehicles, since if the vehicle topples while cornering; it will be because of the faulty suspension design. But in case of an ATV, there are chances that the vehicle will topple while encountering a treacherous terrain.

- The modal analysis was carried out without any consideration of damping components such as vibration isolators, Panels, etc. If they are included, the frequency will be even much lower.

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